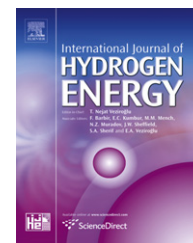


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Performance analysis of hybrid systems incorporating high temperature fuel cells and closed cycle heat engines at part-load operation

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ABSTRACT

This work presents a comparative analysis of hybrid systems based on molten carbonate fuel cells and making use of closed-cycle externally-heated bottoming systems. Two options are considered: reciprocating (Stirling) engines and supercritical carbon dioxide turbines. These two engines share the common feature of working on closed cycles with pure fluids (H_2 and CO_2 respectively) but, at the same time, they differ in their internal structure: Stirling engines make use of volumetric machinery whereas the supercritical carbon dioxide gas turbine (SCO_2) is composed by turbomachinery. In both cases, the working fluid is subjected to very high pressures and temperatures in the range of 50–200 bar and 40–650 °C.

A brief description of both bottoming systems is provided in the first sections of this article along with their expected performance in on-design and off-design (part-load) operation. The analysis of each system is therefore split into three stages. First, the most relevant features of the models of performance are discussed. Then, a comparison is shown for on-design operation aiming to evaluate the maximum efficiency attainable by the proposed engines. Finally, an analysis of off-design operation is presented.

The final section of paper concludes that hybrid systems based on atmospheric fuel cells and externally heated closed-cycle bottoming engines have the potential to outperform conventional pressurised fuel cell plus gas turbine hybrids while avoiding the demanding operating conditions on the fuel cell topping system of the latter configuration. Copyright © 2012, Hydrogen Energy Publications, LLC. Published by Elsevier Ltd. All rights reserved.

1. Introduction

The aim of this work is to develop a new concept of hybrid system based on topping molten carbonate fuel cells and bottoming externally-heated engines. The integration is of the indirect type (i.e. there is heat exchange but no mass transfer)

which brings about a longer life expectancy of the fuel cell due to its smooth atmospheric operation in steady conditions. Ease of operation (load control) and the possibility to run the fuel cell in stand-alone mode are other interesting features.

The first heat engine considered is a closed-cycle gas turbine operating with supercritical carbon dioxide (SCO_2). This is

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a configuration studied extensively so far by the authors, who report a better performance of such system with respect to conventional hybrids (with either indirect or direct integration) in on-design and also off-design conditions [1,2]. The advantage of supercritical fluids is the very low work required to elevate the fluid's pressure at the compressor and hence the higher useful work of the system. The interest of carbon dioxide lies in the close to ambient critical temperature (30.98 °C) and low critical pressure (73.8 bar), which make it possible to develop supercritical cycles with conventional cooling techniques and economical piping.

The second heat engine under consideration is a Stirling engine. It is also a closed-cycle engine but makes use of a reciprocating configuration. The candidate working fluids of a Stirling engine are hydrogen, helium, nitrogen or simply air; in the present paper, hydrogen is selected for it yields the best performance [3].

A Stirling engine comprises: (i) a pair of cylinders where the fluid is compressed at low temperature and expanded at high temperature, (ii) a heater and a cooler to add and reject heat to and from the engine respectively, (iii) a regenerator located in the middle of these heat exchangers in order to increase the efficiency. The main advantages of this engine are compactness, low vibrations, low noise and good part-load performance. Durability and efficiency are also to be expected to depend on the design operating conditions [4].

The benefits of incorporating either of these two engines into a high temperature fuel cell are discussed in this paper in a structured manner. First, conceptual overviews of the reference fuel cell and hybrid systems are presented. Then, the two bottoming systems under consideration are described focusing on the modelling assumptions employed to estimate their performances. Finally, the efficiency in on-design and off-design operation is assessed, showing an important gain when supercritical CO₂ turbines are used instead of the conventional hot air turbines.

2. Basic layout of the reference hybrid system

A general layout of the reference hybrid system is shown in Fig. 1 where a Molten Carbonate Fuel Cell is fed with preheated air and a mixture of water steam and partly-reformed natural gas. This raw fuel is reformed internally in order to take advantage of the heat released by the oxidation of hydrogen. The steam necessary for this process is generated outside the cell in a dedicated heat recovery steam generator that recuperates low

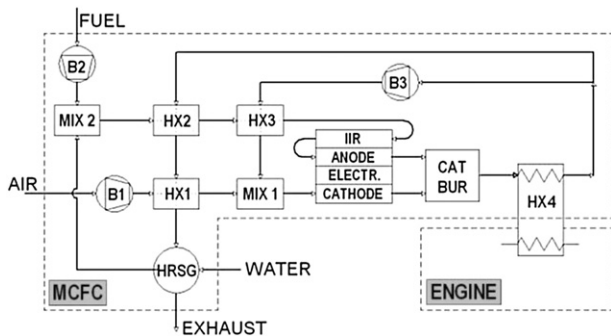


Fig. 1 – Hybrid system layout.

temperature waste heat from the system. This option is preferred to recirculating a fraction of the cell exhaust gases because it allows reducing the stack temperature (hence increasing global efficiency) by optimising the temperature levels at which heat is exchanged by different subsystems of the plant. In other words, high temperature heat is used to drive the bottoming system whereas steam is generated at lower temperature.

A catalytic burner is located downstream the cell exhaust to burn the inevitable excess fuel and increase the temperature of the heat source used to operate the bottoming engine (let it be noted that a minimum temperature of 700 °C is required if the heat engine is to achieve high efficiency). These hot gases enter a heat exchanger in which heat is transferred to the bottoming system (HX4 in Fig. 1) and, then, a fraction of them is recirculated to provide the cathode of the cell with the carbon dioxide necessary to avoid carbonate starvation in the electrolyte. The remaining gas is used to preheat the fuel and air streams into the cell and to generate steam for the internal reformer.

3. Models of performance

3.1. Fuel cell

This work makes use of a lumped-volume model of performance of molten carbonate fuel cells with indirect internal reforming, of which a very detailed discussion is available in reference [5] by the authors. The fundamental equations that govern the operation of the fuel cell are listed below:

- Nernst voltage equation: ideal open circuit potential depending on the pressure, temperature and composition of the reactants. Compositions are averaged logarithmically between the inlet and outlet sections of the cell.

$$E_{\text{Nernst}} = -\frac{\Delta G_f}{2 \cdot F} = -\frac{\Delta G_f^0}{2 \cdot F} + \frac{R_u \cdot T}{2 \cdot F} \cdot \ln \left(\frac{p_{\text{H}_2} \cdot p_{\text{O}_2}^{1/2}}{p_{\text{H}_2\text{O}}} \right) + \frac{R_u \cdot T}{2 \cdot F} \cdot \ln \left(\frac{p_{\text{CO}_2, \text{an}}}{p_{\text{CO}_2, \text{cat}}} \right) \quad (1)$$

- Ohmic resistance: internal resistance to the flow of electronic/ionic charge within electrodes/electrolyte.

$$R_{\text{ir}} = A_{\text{ir}} \cdot \exp(\Delta H_{\text{ir}} / (R_u \cdot T)) \quad (2)$$

- Activation losses at the anode: responsible for the voltage drop due to the need to overcome the activation potential of the oxidation half-reactions.

$$R_{\text{an}} = A_{\text{an}} \cdot p_{\text{H}_2}^{-0.5} \cdot \exp(\Delta H_{\text{an}} / (R_u \cdot T)) \quad (3)$$

- Activation and concentration losses at the cathode: the first term is conceptually similar to Eq. (3) whereas the second takes into account the rate of consumption of carbon dioxide to form carbonate ions at this electrode ($\text{CO}_2 + \text{O}_2 + 2e^- \rightarrow \text{CO}_3^-$).

$$R_{\text{ca}} = A_{\text{ca},1} \cdot p_{\text{O}_2}^{-0.75} \cdot p_{\text{CO}_2}^{0.5} \cdot \exp(\Delta H_{\text{ca},1} / (R_u \cdot T)) + A_{\text{ca},2} \cdot x_{\text{CO}_2}^{-1.0} \cdot \exp(\Delta H_{\text{ca},2} / (R_u \cdot T)) \quad (4)$$

- Operating current/voltage of the cell: Faraday's law is used to evaluate the rate of consumption of hydrogen. This

information is then combined with the definition of fuel and carbon utilisations (U_f and U_{CO_2}) to determine the (raw) fuel flow rate and the rate of exhaust gas recirculation to the cathode inlet respectively. For the fuel, a standard hydrocarbon fuel $C_\alpha H_\beta$ is considered, yielding the definition of the steam-to-carbon ratio given in Eq. (9). This parameter determines the amount of steam that has to be supplied to the reforming process to promote the conversion of hydrocarbons to hydrogen rather than the formation of elementary carbon. It takes values in the range from 2 (the risk of carbonisation increases) to 5 (voltage drops due to the dilution of reactants at the anode).

$$V = E_{Nernst} - (R_{TOTAL}) \cdot j \quad (5)$$

$$U_f = \left(\frac{j \cdot A}{2 \cdot F} \right) / \left(x_{CO} \cdot \dot{n}_{fuel} + \sum_{\alpha=1}^4 \left(2 \cdot \alpha + \frac{1}{2} \cdot \beta \right) \cdot x_{C_\alpha H_\beta} \cdot \dot{n}_{fuel} \right) \quad (6)$$

$$U_{CO_2} = \left(\frac{j \cdot A}{2 \cdot F} \right) / \dot{n}_{CO_2, cathode \text{ inlet}} \quad (7)$$

$$\bar{p}_i = (p_i^{in} - p_i^{out}) / \ln(p_i^{in} / p_i^{out}) \quad (8)$$

$$STCR = \frac{\dot{n}_{H_2O}}{\dot{n}_{CO} + \dot{n}_{CO_2} + \sum \dot{n}_{C_\alpha H_\beta} \cdot \alpha} \quad (9)$$

3.2. Supercritical carbon dioxide turbine

The supercritical carbon dioxide power cycle has been extensively reviewed in literature in the last few years, starting from Dostal's review [6] of the works by Angelino and Feher [7,8]. This growing interest has given way to dedicated technical meetings (Supercritical CO₂ Power Cycle Symposium held once every two years in US) and a number of research works, most of which focus on nuclear power generation in high temperature gas reactors. More recently, SCO₂ systems have been proposed for solar applications [9] and hybrid systems with fuel cells [10].

Developing a model of performance of closed-cycle gas turbines involves the analysis of two types of components: turbomachinery (compressor and turbine) and heat exchangers (recuperator, heater and condenser/cooler). So far, different authors have approached the simulation of this turbomachinery based on the well founded thermodynamic and aerodynamic principles that are commonly used to design air compressors and gas or steam turbines [11,12]. Nevertheless, this is not correct when applied to supercritical CO₂ systems as noted by some works in the subject [13,14]. The markedly non-ideal behaviour of carbon dioxide near the critical point brings about the need to develop a specific database on which an efficient design of the compressor (and to a lesser extent the turbine) can rely. This is better assessed in Table 1 where the compressibility factors at the most relevant stations of the cycle are summarised for air and supercritical carbon dioxide.⁴

⁴ The compressibility factor Z is an index to evaluate how close to ideal the behaviour of a particular gas at a certain pressure and temperature is: $p \cdot v = Z \cdot R \cdot T$. Thus, $Z = 1$ for ideal gases.

Table 1 – Compressibility factor Z for air and supercritical CO₂ at the inlet/outlet to/from the compressor/turbine.

Location	Variable	Air	S-CO ₂
Compressor inlet	T_r/P_r	2.259/0.027	1.014/1.017
	Z	1	0.25
Compressor outlet	T_r/P_r	3.294/0.080	1.271/3.050
	Z	1	0.60
Turbine inlet	T_r/P_r	6.994/0.077	3.037/2.929
	Z	1	1.02
Turbine outlet	T_r/P_r	5.588/0.027	2.609/1.059
	Z	1	1

This scenario suggests that a generalised performance map be employed in lieu of the analytical (equation-based) models typically used to estimate the performance of compressors in gas turbine engines. In order to have as accurate as possible estimations of the compressor performance, this map is adapted from the available experimental information gathered at the supercritical carbon dioxide test rig at Sandia National Laboratories, where a centrifugal compressor running on CO₂ at supercritical inlet conditions was tested in 2009 and 2010 [15]. For the turbine, the vanes are assumed to be choked and hence a single equivalent-nozzle curve is matched to the compressor map to yield the running line of the system in a pressure ratio vs corrected mass flow plot. The outcome of this approach is illustrated in Figs. 2 and 3 where the corrected mass flow rate in the compressor and turbine is defined as follows:

$$\dot{m}_c = \frac{\dot{m}}{\rho_{0,in} \cdot a_{0,in}} \quad (10)$$

$$\dot{m}_t = \frac{\dot{m} \sqrt{T_{0,in}}}{P_{0,in}} \quad (11)$$

Note that subscript $_{0,in}$ in Eqs. (10) and (11) indicates stagnation conditions at the compressor/turbine inlet. It is also worth noting that the corrected mass flow is expressed differently in compressor and turbine, due to the different behaviour of the fluid at the inlet to each component. In effect,

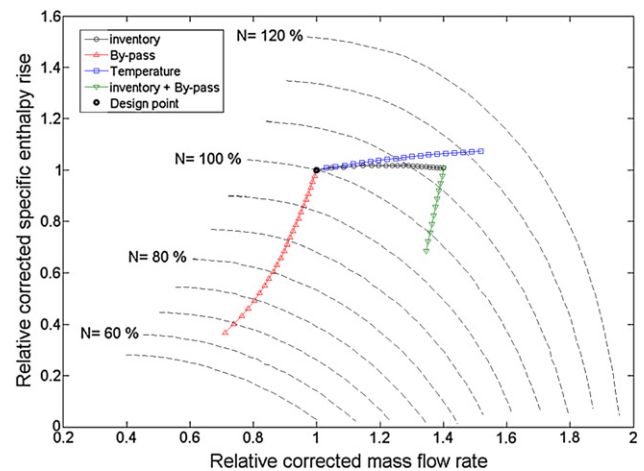


Fig. 2 – SCO₂ compressor performance map [15].

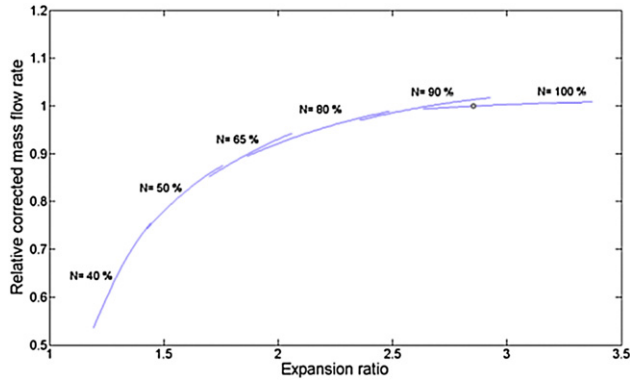


Fig. 3 – SCO₂ turbine performance map [16].

Table 1 shows that carbon dioxide behaves close to ideally at the turbine (compressibility factor close to unity) whereas strong real gas effects are experienced at the compressor (compressibility factor ranging from 0.25 to 0.60). This latter feature of the fluid is accounted for by introducing R , γ and ρ in the definition of $\dot{m}_c$ as deduced from the application of Buckingham's Π theorem [16].

Heat exchangers are modelled in this work by means of an ε -NTU approach where effectiveness ε and number of transfer units NTU are defined as follows:

$$C^* = C_{\min}/C_{\max} = (\dot{m} \cdot c_p)_{\min}/(\dot{m} \cdot c_p)_{\max} \leq 1 \quad (12)$$

$$\varepsilon = \frac{Q}{Q_{\max}} = \frac{C_{\text{hot}}(T_{\text{hot,in}} - T_{\text{hot,out}})}{C_{\min}(T_{\text{hot,in}} - T_{\text{cold,in}})} = \frac{C_{\text{cold}}(T_{\text{cold,out}} - T_{\text{cold,in}})}{C_{\min}(T_{\text{hot,in}} - T_{\text{cold,in}})} \quad (13)$$

$$\text{NTU} = A \cdot U / C_{\min} \quad (14)$$

C^* in Eq. (12) is the heat capacity ratio where the heat capacity C of a fluid is defined as the product of mass flow times the specific heat at constant pressure. T stands for temperature of the cold (c) and hot (h) fluids at the inlet (in) and outlet (out) sections of the heat exchangers. Based on Eqs. (12) and (13), a specific correlation between ε and NTU exists depending on the type and internal flow arrangement of the equipment. For a cross flow PFHE or PCHE, this equation is:

$$\varepsilon = \frac{1 - \exp(-(1 - C^*)\text{NTU})}{1 - C^* \exp(-(1 - C^*)\text{NTU})} \quad (15)$$

The combination of Eqs. (14) and (15) provides an estimation of the global heat transfer coefficient and exchange area of the component. A more detailed description of the calculation procedure is available in reference [1].

Off-design operation of the gas turbine is performed by a combined system based on inventory and by-pass controls. At high loads, the output of the turbine is controlled by modifying the circulating mass flow in the system whilst the turbine inlet temperature remains constant. This is accomplished by charging and discharging an auxiliary storage vessel connected in parallel to the compressor (Fig. 4). Nevertheless, this control strategy brings about an acceleration of the shaft at reduced loads which finds a limit at a certain rotating speed for which the centrifugal forces acting on the blade tips become too high. For further load reductions,

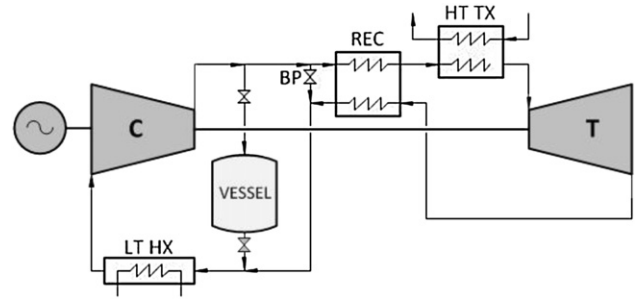


Fig. 4 – Gas turbine layout showing control subsystems.

a secondary control strategy is required which, in this case, is a turbine by-pass valve. This secondary control strategy consists on reducing the expanding mass-flow at the turbine while keeping it constant at the compressor. The net shaft work is so reduced.

It has to be noted here that other control strategies like modifying the turbine inlet temperature are also possible and even yield fairly high efficiencies as shown in Fig. 5. Furthermore, the comparison shown in this latter figure confirms that, apart from inventory control, the cycle is more efficient when a temperature control is adopted as opposed to using a turbine by-pass valve. Nevertheless, in spite of this observation, using an inventory plus turbine inlet temperature combined control system is not feasible in practice since both strategies result in higher shaft speeds at reduced loads. That is to say, inventory control is limited by the highest shaft speed that the impeller/rotor tolerates. If a turbine inlet temperature were adopted for further load reduction, the shaft would speed-up bringing about a mechanical failure of the rotating parts of the CO₂ system. It must be remarked that this is because of the very high sensitivity of compressor

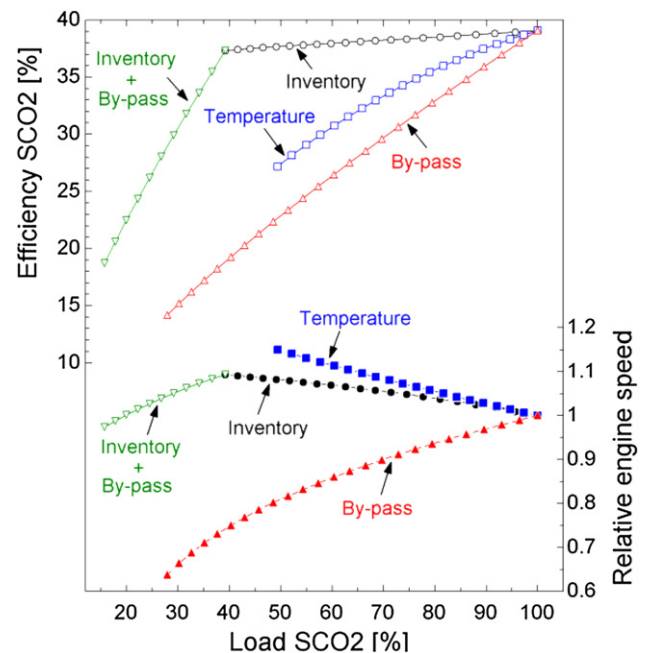


Fig. 5 – Part load performance of the SCO₂ cycle.

performance to inlet density (ρ_{01}) which varies dramatically when the low pressure level (p_{01}) of the cycle is reduced at part-load. Moreover, Eq. (13) shows that a density reduction would inevitably lead to higher corrected flow rates and, consequently, higher rotating speeds. Eventually, the maximum shaft speed for safe operation would be exceeded and the turbine would fail. Hence, an inventory followed by turbine by-pass system is most convenient.

In light of the results above, the following control system is recommended as shown in Fig. 5:

- 100–50% load range: inventory control. The lower limit of 50% is set as the load for which the shaft rotates at 110% its rated speed.
- Loads lower than 50%: by-pass control. The shaft speed reduces to about 80% of its rated value at 15% output.

3.3. Stirling engine

The working cycle of a Stirling engine shares common features with that of the SCO_2 turbine since both are based on compressing a gas when it is cold and expanding it when it is hot. Nonetheless, the reciprocating layout of Stirling engines incorporates differential aspects as shown in Fig. 6 where the working cycle of these systems is depicted. The area enclosed by the p – V diagram represents the indicated (thermodynamic) work per cycle developed by a pair of cylinders:

$$W_{\text{ind}} = \oint P dV \quad (16)$$

The models of performance of Stirling engines are usually classified in categories depending on their complexity: zeroth order (experimental curve fitting), first order (rough estimations based on simplified models), second order (local effects like pressure drop or heat transfer are incorporated), third order (conservation of mass, momentum and energy applied to elements of the fluid domain within the engine) and fourth order (computational fluid dynamics applied). In this paper, a third order model of performance is used whose complete details are given in Ref. [17].

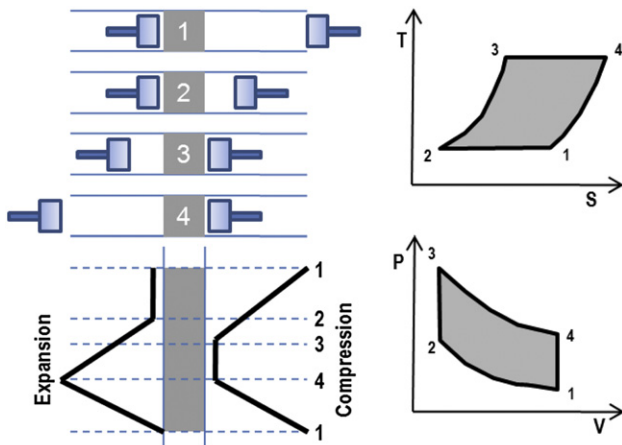


Fig. 6 – Principle of operation of a Stirling engine.

The reference engine is an alpha-type squared-four unit with double-acting cylinders, in which a 90° angle delay between each pair of compression-expansion volumes makes it possible to develop four working cycles in each crankshaft turn. One such pair of compression/expansion cylinders is shown in Fig. 7 where the internal geometry of the engine is clearly illustrated: compression cylinder (C), cooler (K), regenerator (R), heater (H) and expansion volume (E); inter-connection volumes between adjacent components are also accounted for in the calculations.

The model is based on solving the equations of conservation of mass, momentum and energy at each volume (node) shown in Fig. 7:

- Mass conservation:

$$dm/dt = \dot{m}_{\text{in}} - \dot{m}_{\text{out}} \quad (17)$$

- Energy conservation:

$$\frac{d(m \cdot u)}{dt} = \dot{Q} + \dot{m}_{\text{in}} \cdot (h_{\text{in}} + 0.5 \cdot v_{\text{in}}^2) - \dot{m}_{\text{out}} \cdot (h_{\text{out}} + 0.5 \cdot v_{\text{out}}^2) - \dot{W} \quad (18)$$

- Momentum conservation:

$$v \cdot dv + \frac{f \cdot v^2}{2 \cdot d_h} \cdot dx + \frac{dp}{\rho} = 0 \quad (19)$$

The previous equations are complemented by the following equation of state for real gases [18]. The utilisation of this equation makes it possible to account for the dense gas effects of hydrogen at high pressure and low temperature as follows ($k = 1.9 \times 10^{-6}$ for hydrogen):

$$P_i \cdot V_i = m_i \cdot R_u \cdot (T_i + k \cdot P_i) \quad (20)$$

Starting from the principle of mass conservation applied to the engine as a whole, Eq. (21), and incorporating the equation of state just shown, Eq. (20), a codable equation to calculate

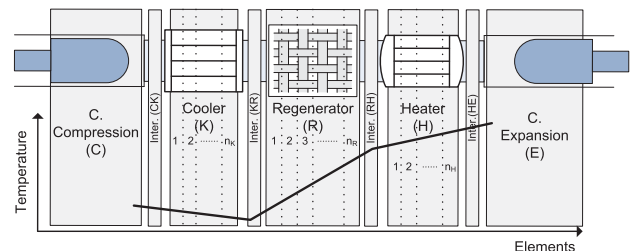


Fig. 7 – Calculation domain for the reference engine (pair of cylinders). Note local refinement of heat exchangers.

the reference pressure in the compression cylinder at time t is obtained, Eq. (22):

$$m_C + m_{CK} + \sum_{i=1}^{n_k} m_{K_i} + m_{KR} + \sum_{i=1}^{n_r} m_{R_i} + m_{RH} + \sum_{i=1}^{n_h} m_{H_i} + m_{HE} + m_E = M \quad (21)$$

$$P_C^t = \frac{M \cdot R_u}{\frac{V_C^t}{T_C^t + k \cdot p_C^t} + \frac{F_{CK}^t \cdot V_{CK}}{T_{CK}^t + k \cdot p_{CK}^t} + \sum_{i=1}^{n_k} \left(\frac{F_{K_i}^t \cdot V_{K_i}}{T_{K_i}^t + k \cdot p_{K_i}^t} \right) + \frac{F_{KR}^t \cdot V_{KR}}{T_{KR}^t + k \cdot p_{KR}^t} + \sum_{i=1}^{n_r} \left(\frac{F_{R_i}^t \cdot V_{R_i}}{T_{R_i}^t + k \cdot p_{R_i}^t} \right) + \frac{F_{RH}^t \cdot V_{RH}}{T_{RH}^t + k \cdot p_{RH}^t} + \sum_{i=1}^{n_h} \left(\frac{F_{H_i}^t \cdot V_{H_i}}{T_{H_i}^t + k \cdot p_{H_i}^t} \right) + \frac{F_{HE}^t \cdot V_{CK}}{T_{HE}^t + k \cdot p_{HE}^t} + \frac{F_E^t \cdot V_E^t}{T_E^t + k \cdot p_E^t}} \quad (22)$$

where F_i is the ratio from the pressure in volume i to the reference pressure:

$$F_i = P_i / P_C \quad (23)$$

An iterative process is implemented in order to calculate the pressure inside the compression and expansion cylinders at each time step (and therefore the free volume inside the engine); the P – V diagram (and therefore the indicated work) is so obtained. The following assumptions are incorporated in the model:

- For the calculation of pressure losses:
 - This effect is coupled to the thermodynamic model (i.e. it is not independent as it is in second order models).
 - Inertial losses and oscillatory flow effects are accounted for in the regenerator. On the contrary, the latter effects are neglected in circular pipes (heater/cooler) due to the high “fluid displacement/pipe diameter” ratio; stationary flow correlations are used instead [19].
 - Pressure losses are also considered in the interconnection volumes.
- For the heat transfer calculations:
 - Conductive heat transfer is accounted for within the gas (stream-wise) and between adjacent elements of the cooler, regenerator and heater walls.
 - Oscillatory flow correlations are used to evaluate Nusselt number in the regenerator whereas stationary flow correlations are used in circular pipes.

The model so developed has been validated against experimental data available at the NASA Lewis Research Centre [20] showing a more than satisfactory agreement [17].

4. Hybrid system performance

4.1. On-design operation

The performances of hybrid systems comprising a topping molten carbonate fuel cell and either a bottoming supercritical carbon dioxide turbine or a bottoming Stirling engine are

now studied. To this aim, a reference 500 kWe fuel cell is considered with the operating conditions summarised in Table 2.

The design conditions of the supercritical carbon dioxide turbine are shown in Table 3. The most relevant information concerns the thermodynamic state of the fluid at the inlet to

the compressor, 308 K and 75 bar, which is very close to the critical point. For the specifications of turbomachinery and heat exchangers, these have been taken from an exhaustive literature review as reported by the authors in Ref. [2].

Finally, the rated operating conditions of the Stirling engine are listed in Table 4 where novel concepts to indicate the performance of the engine are needed due to its reciprocating nature.⁵ These parameters are the mean effective pressure (mep) and the mean pressure for mechanical losses (mpml). The mean effective pressure results from dividing the work developed (by a pair of cylinders) in a cycle by the volume displaced in the same period of time.⁶ Similarly, the mean pressure for mechanical losses results from dividing the mechanical losses (experienced by a pair of cylinders) by the volume displaced in a cycle. These parameters are commonly used in the field of reciprocating engines and their mathematical descriptions are given below [21]:

$$\text{mep} = \left(\oint P dV \right) / V_D \quad (24)$$

$$\text{mpml} = W_{\text{loss}} / V_D \quad (25)$$

A comparison of Tables 3 and 4 yields an interesting conclusion which is the higher efficiency of the supercritical carbon dioxide turbine when the same maximum temperatures are considered; this difference is in the order of 8–9%. Nevertheless, in spite of the lower efficiency of the Stirling engine, it is also remarkable that this latter engine is far more efficient than the conventional hot air turbine still proposed in many hybrid systems using atmospheric MCFCs. More in particular, the expected efficiency of such a system is around 27%, as shown in Ref. [2] by the authors, what means that making use of a Stirling engine would provide a 30% gain in bottoming system efficiency.

⁵ Note that the crankshaft mechanism impelling the flow of hydrogen from one cylinder to another makes it impossible to define a rated continuous flow like in a fuel cell or gas turbine engine.

⁶ The mean effective pressure is a virtual pressure which acting on the piston from the top dead centre to the bottom dead centre (a complete stroke) develops the same effective work as the engine does in a cycle.

Table 2 – Fuel cell rated performance.

Parameter	Value
Current density [A m^{-2}]	1100
Voltage [V]	0.716
Anode/cathode pressure loss [%]	1/2
Fuel flow [g s^{-1}]	19.9
STCR [–]	3
Temperature [K]	923
Fuel/ CO_2 utilization [%]	77.4/70
Gross power output [kWe]	500
Auxiliary power [kWe]	52.4
Net efficiency [%]	47.6

These aspects are illustrated in Table 5 which provides a comparison between the rated performance of hybrid systems using either hot air turbine (standard case in the public domain), supercritical CO_2 turbines or Stirling engines. The following issues are to be noted:

- The efficiency gain when a supercritical CO_2 turbine is used is remarkable. In particular, for the same waste heat available from the topping system (MCFC), a 50% increase in output from the bottoming system is attained with respect to the same configuration using a hot air turbine. In terms of global power production, the previous effect has an impact of more than 5%.
- The share of power delivered by the bottoming system when the SCO_2 turbine is used approaches 20%.
- Using a bottoming Stirling engine in lieu of a hot air turbine yields a performance lying between that of a hot air turbine and a supercritical CO_2 system, even if it is closer to the first case.

Another interesting aspect to be noted in Table 5 is the peak temperature of the Stirling engine, which is quoted as 910 K. Given the volumetric nature of the system, the maximum temperature of the working gas within the engine is not constant, as opposed to the other turbine-based cases. Actually, this temperature depends on the volumetric flow of gases through the heater and on whether these gases flow from the compression cylinder to the expansion volume or vice versa. As a result, the peak temperature achieved by the working gas varies non-linearly in the range from 780 to 935 K,

Table 3 – Rated performance of the SCO_2 cycle.

Parameter	Value
Regenerator pressure loss hot/cold side [%]	1.5/0.5
Heater pressure loss [%]	2
Precooler pressure loss [%]	1
Compressor inlet pressure/temperature [bar/K]	75/308
Pressure ratio [–]	3:1
Turbine inlet temperature [K]	923
Compressor/turbine efficiency [%]	90/95
Recuperator effectiveness [%]	95
Mechanical efficiency [%]	98
Shaft work [kJ kg^{-1}]	107.7
Gross efficiency [%]	39.1

as a function of the crank-angle. Thus, an average value of 910 K is indicated in Table 5.

The performance of the system in part-load operation is illustrated in Fig. 8 which confirms that the utilisation of externally-heated closed-cycle bottoming systems is beneficial in terms of system efficiency. In particular, an efficiency gain of around six percentages is to be expected for a 50% load reduction in both cases (58–65% for the SCO_2 systems and 56 to 61 for the Stirling-based hybrid). The root cause for this behaviour is found in the superb part-load performance of the bottoming cycles whose efficiencies remain fairly constant even at reduced load settings. This is particularly evident in the case of the supercritical CO_2 turbine, which exhibits a drop in efficiency from roughly 40% to around 37% in the aforementioned range.

The information provided in Table 5 and Fig. 8 confirms that the integration of molten carbonate fuel cells and closed-cycle heat engines has the potential to boost the efficiency of atmospheric hybrid systems to values in the order of 60% while, at the same time, preserving the safe operating conditions of the cell. This is thanks to the variety of part-load mechanisms available for closed cycles, which yield a comparative advantage over conventional systems where temperature and shaft-speed control only are possible. Even though such control strategies demand for more complex subsystems (for instance inventory tanks or additional valves), the increased power output or reduced fuel consumption and the already complex layout of an internal reforming fuel cell system justifies the utilisation of the proposed configuration.

Finally, some additional remarks must be highlighted with regard to the choice of supercritical carbon dioxide turbines or Stirling engines. First, despite the higher efficiency of the SCO_2 option, Stirling engines are in a more advanced state of development though it must be acknowledge that they still lack economic competitiveness due to the very high manufacturing and maintenance costs. In this regard, it would be possible to have pilot MCFC-Stirling pilot plants installed in the short term. In contrast, SCO_2 turbines are still an immature technology as deduced from the very few experimental tests in operation which, in most cases, concentrate on one aspect of the system or on demonstrating its technical feasibility rather than on developing a product to

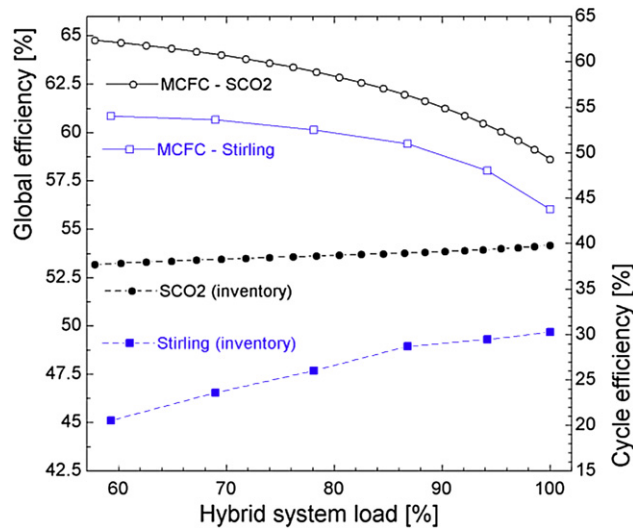
Table 4 – Rated performance of the Stirling engine.^a

Parameter	Value
Mean pressure [bar]	150
Shaft speed [rpm]	1750
Heater wall temperature [K]	993
Cooling water inlet temperature [K]	323
Effective work [kJ cycle^{-1}]	976.5
Mean effective pressure [bar]	18.6
Mean pressure for mechanical losses [bar]	1.1
Gross efficiency [%]	36.07

^a Note that even though the effective work per cycle is given for the reference engine used for validation (30–35 kWe rated power output), the mean effective pressure has the noteworthy advantage of not being affected by the engine displacement. The same applies to the mean pressure for mechanical losses.

Table 5 – Rated performance of the hybrid systems.

Parameter	Air	SCO ₂	Stirling
Effective power MCFC [kW]	447.6		
Effective power heat engine [kW]	71.7	103.5	79.8
Global effective power [kW]	519.2	551.1	526.7
Ratio Power MCFC/Engine [%]	86.2/13.8	81.2/18.8	85.0/15.0
Efficiency MCFC [%]	47.6		
Engine peak temperature [K]	948.6	947.1	910
Net efficiency bottoming engine [%]	27.6	39.8	30.3
Global net efficiency [%]	55.2	58.7	56.0

**Fig. 8 – Part-load efficiency of the hybrid systems under consideration.**

be deployed to the market. It is hence expected that these systems approach the commercial stage in the mid-term.

Further to these considerations, another differential aspect between both bottoming systems is the output range. Hence, Stirling engines are commercially available in the range of tens of kilowatts (<100 kW), with some companies reporting to have plans to develop units able to deliver hundreds of kilowatts (200–500 kW) [22], whereas the supercritical carbon dioxide turbine is expected to be cost-effective for power outputs approaching 1 MW and beyond [23]. In light of these considerations, Stirling engines are most convenient for small-scale hybrid systems whilst SCO₂ turbines should be adopted in mid and large-scale hybrid power plants.

5. Conclusions

This paper presents a comprehensive assessment of the most interesting bottoming system to be used in hybrid systems composed by high temperature molten carbonate fuel cells and bottoming heat engines of the closed cycle type. The rationale behind this analysis is to explore the benefits that could eventually be derived from the flexible features of closed power cycles as opposed to open systems: working fluid, system pressure, part-load strategy and others. Two candidate engines are considered: Stirling engines

(volumetric) and closed-cycle turbines operating on supercritical carbon dioxide.

The main conclusions drawn from this work are:

- Comprehensive and accurate modelling tools are provided for each subsystem (fuel cell, Stirling engines and SCO₂ turbine), with the capacity to simulate the performance in on-design and off-design operation of each one of them.
- The utilisation of a supercritical carbon dioxide turbine is expected to boost the efficiency of the corresponding hybrid system to values in the order of 60% for systems comprising fuel cells operating at atmospheric pressure.
- Using a Stirling engine is somewhat less interesting due to the limited efficiency gain with respect to standard hybrid systems that incorporate hot air turbines. The main reason for this is the effect of the volumetric nature of the unit on the peak temperature achieved by the working gas. This effect brings about a reduction in engine efficiency.
- The efficiency of the SCO₂ turbine remains almost unaltered in a wide range of load-settings, hence providing a remarkable part-load performance of the corresponding hybrid system.
- This pattern is somehow observed in the Stirling-based configuration though the performance of this system is worse.
- SCO₂ turbines and Stirling engines fit to different output ranges. Hence, Stirling engines are suitable for outputs lower than a hundred kilowatts whilst SCO₂ turbines are most interesting for outputs in the range of a few hundred kilowatts or more.

Further research work by the authors is aimed at assessing the costs of both systems and the potential to benefit from economies of scale, even if such an analysis incorporates important uncertainty. Once this complementary information is available, the technical comparison provided in this paper would be integrated with the economic information in order to identify the most cost-effective technology to be used in the next generation of hybrid systems.

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- C_p : specific heat, $J\ kg^{-1}\ K^{-1}$
 d_h : hydraulic diameter, m
 E : Nernst voltage, V
 f : friction factor, –
 F : Faraday constant, 96,485 C
 G : Gibbs free energy, $J\ kg^{-1}$
 h : enthalpy, $J\ kg^{-1}$
 j : current density, $A\ m^{-2}$
 K : equilibrium constant, –
 m : mass, kg
 mep : mean effective pressure, bar
 $mpml$: mean pressure for mechanical losses, bar
 $\dot{m}$: mass flow, $kg\ s^{-1}$
 M : total mass, kg
 $MCFC$: molten carbonate fuel cell
 NTU : number of transfer units, –
 $\dot{n}$: molar flow, $mol\ s^{-1}$
 p : partial pressure, bar
 P : pressure, bar
 $\dot{Q}$: heat, W
 R_u : universal gas constant, $8.3145\ J\ mol^{-1}\ K^{-1}$
 R : resistance, $\Omega\ m^2$
 SCO_2 : supercritical carbon dioxide turbine
 $STCR$: steam to carbon ratio, –
 t : time, s
 T : temperature, K
 u : internal energy, $J\ kg^{-1}$
 U : overall heat transfer coefficient, $W\ m^{-2}\ K^{-1}$
 U_{CO_2} : carbon utilisation factor, %
 U_f : fuel utilization, –
 v : velocity, $m\ s^{-1}$
 V : voltage, V
 V_i : volume, m^3
 W : work, $J\ kg^{-1}$
 $\dot{W}$: power, W
 x : molar fraction, –
 Z : polarisation, V
- Greek**
 ϵ : effectiveness, %
 ρ : density, $kg\ m^{-3}$
- Subscripts**
 an : anode
 C : compression cylinder
 ca : cathode
 CK : connecting duct compression cylinder–cooler
 D : displaced
 E : expansion cylinder
 H : heater
 HE : connecting duct heater–expansion cylinder
 in : inlet
 ir : Ohmic
 K : cooler
 KR : connecting duct cooler–regenerator
 max : maximum
 min : minimum
 out : outlet
 R : regenerator
 RH : connecting duct regenerator–heater

Glossary

Latin

a : sound speed, $m\ s^{-1}$
 A : area, m^2
 C : heat capacity, $W\ K^{-1}$